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THE PERFORMANCE PREDICTION OF A 30-KW MULTI-FUELED MICRO GAS TURBINE

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ABSTRACT

This study focuses on the performance prediction of a 30-kW micro gas turbine, using natural gas, ethanol from sugarcane, and biogas from palm-oil residues. The theories of thermodynamics, fluid mechanics, and mathematical modelling were applied. To clarify the full-load and part-load performance of the micro gas turbine, a compressor performance characteristic map and a turbine performance characteristic map were developed. A steady-state thermodynamic model of the micro gas turbine was developed and validated against the manufacturing data for a Capstone C30 micro gas turbine, the results from testing a Capstone C30, and the analytical results. Natural gas properties were applied for model-validation purposes and the results were found to be in good agreement. The effects of the fuel variation on the electrical efficiency, fuel consumption, heat recovery at the recuperator, exhaust temperature, and power output of the micro gas turbine were investigated. This study found no significant difference in the micro-gas-turbine performance when the fuel was changed from natural gas to renewable energy; i.e., ethanol and biogas. However, utilizing renewable energy instead of natural gas required much more fuel to be fed into the system, which increased the fuel expenses and operational costs.

***KEYWORDS:** Micro Gas Turbine, Compressor, Turbine, Ethanol, Biogas & Part-Load Performance*

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Original Article

1. INTRODUCTION

Micro gas turbines are a relatively new distributed-generation technology with an output range of 20 to 500 kW. Micro gas turbines are generally used in distributed-generation applications because of their ability to operate in both stand-alone and grid-connected modes [1–3]. Moreover, micro gas turbines have proven capable of operating at high speeds (about 43,000 to 96,000 revolutions per minute) compared with the larger gas turbines, due to their single-shaft arrangement. Nowadays, the thermal efficiencies of natural-gas-fueled commercial recuperated micro gas turbines from leading micro-gas-turbine manufacturers are around 25% to 33%, by employing recuperators operating at 79% effectiveness [4]. Numerous studies have attempted to analyze the micro gas turbine's performance, operating at full-load and part-load conditions. N. Zhang and R. Cai [5] determined the explicit analytical part-load performance of a constant rotating-speed single-shaft micro gas turbine and its cogeneration. A compressor performance expression and a turbine performance expression were shown in terms of the typical relations with simple analytical functions. The performance expressions, together with an improved Flügel formula, were applied. In a follow-up study, Wang et al. [6] applied a modified version of this calculation strategy to investigate the optimum speed variation for off-design micro-gas-turbine performance. It can be concluded from the study that the micro-gas-turbine performance calculation is strongly affected by the choice of

constants in the compressor and turbine performance formulas. M. Kiaee et al. [7] simulated a commercial single-shaft variable-speed micro gas turbine, the Turbec T100, using a performance adaptation method. Design- and off-design-point adaptation algorithms were investigated. A design-point adaptation algorithm that contains thermodynamic constraints was developed, and characteristic equations for compressors and turbines, in terms of second-order algebraic equations, were presented. A. Gimelli et al. [8] applied a multi-variable multi-objective methodology and the Euclidean norm to a single-stage single-shaft recuperated micro gas turbine, the Capstone C30. A genetic algorithm called MOGA-II and the Pareto optimality concept were applied for model verification. Experimental data from the literature, manufacturer datasheets, and published articles were examined to define the decision variables and objectives of the optimization algorithm. The optimization objective was to maximize the power output and efficiency by varying the turbine outlet temperature and rotational speed. The study discovered that the maximal electrical efficiency could not be reached at the maximal electrical power output. Although commercial micro gas turbines have been designed to use natural gas as their primary fuel, interest is growing in the utilization of biofuels in micro gas turbines. H. N. Somehsaraeietal. [10] investigatedthe influence of changing the fuel from natural gas to biofuel with various methane contents on the Turbec T100 micro gas turbine, operating at different power-load demands. A steady-state thermodynamic model of a micro gas turbine using biogas was developed using IPSEpro, a heat and mass balance software tool, and validated against data obtained from a commercial Turbec T100 micro-gas-turbine test rig. Enhancing the performance of a Turbec T100 micro gas turbine using a fogging inlet air-cooling technique at the compressor inlet was studied [11]. The results indicated that the electric power output of the micro gas turbine linearly increases with the reduction of the air inlet temperature.

Since the micro-gas-turbine performance is altered by a sudden change of load, ambient condition, or fuel type, predicting the micro-gas-turbine performance at full-load and part-load operations is therefore essential for accurate system design in the early preliminary-design phase. The objective of this research paper was to present the performance prediction of a 30-kW micro gas turbine, using various types of fuel, with different electrical-load demands. Natural gas, ethanol from sugarcane, and biogas from palm-oil residues were selected for investigation through out the analysis. The developed micro gas turbine consists of a single-stage centrifugal compressor, a single-stage radial in flow turbine, a combustion chamber, a recuperator, and a generator. The theories of thermodynamics, fluid mechanics, and mathematical modeling were applied to obtain the compressor and turbine characteristic maps. The effects of the fuels on the system's performance are fully described.

2. RESEARCH METHODOLOGY

2.1. Thermodynamics of a 30-kW Micro Gas Turbine

A schematic Brayton-cycle T-s diagram is shown in Figure 1. Natural gas is used as the primary fuel for the preliminary phase of the study. The assumptions are as follows: 1) The micro gas turbine operates as a stand-alone unit. 2) A single-stage centrifugal compressor and radial inflow turbine are mounted on a common shaft. 3) The compression and expansion processes are adiabatic and irreversible. 4) Air is considered a working fluid and its specific heat at constant pressure is assumed as constant throughout the process. 5) The specific heats of air and hot gas are 1.005 kJ/kg K and 1.148 kJ/kg K, respectively.6) The system pressure losses (cool-side) in the micro gas turbine are 6% less than the compressor pressure ratio. 7) The lower heating value of natural gas is approximately 45, 860 kJ/kg.

2.1.1. Centrifugal Compressor

The compressor's pressure ratio is about 3:1 to 5:1. The relationship between the compression ratio and the compressor-absorbed power, based on [12,13], are shown in Eqs. (1)–(3) [14,16].

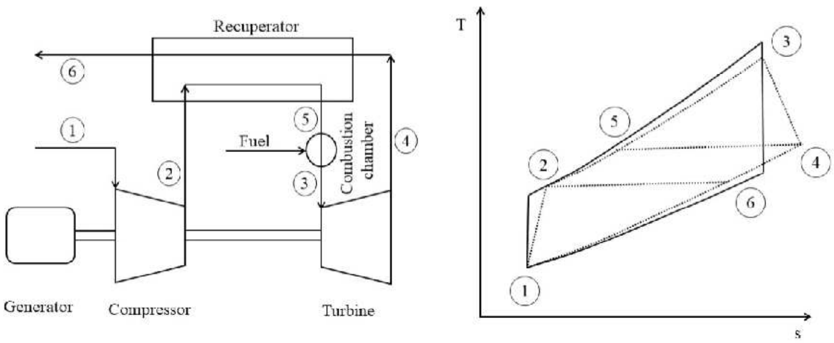

Figure 1: The Micro Gas Turbine Schematic Diagram and Brayton Cycle

$$\pi_c = \frac{P_{02}}{P_{01}} \tag{1}$$

$$\frac{T_{02}}{T_{01}} = \frac{1}{\eta_c}\left(\pi_c^{\frac{\gamma-1}{\gamma}} - 1\right) + 1 \tag{2}$$

$$\dot{w}_c = \dot{m}_a c_{pa} T_{01}\left(\frac{T_{02}}{T_{01}} - 1\right). \tag{3}$$

2.1.2. Combustion Chamber

The lower heating values of the fuel, air mass flow rate, and fuel flow rate [17] are correlated and are expressed in Eqs. (4)–(7).

$$P_{03} = P_{05}\left(1 - \frac{\Delta P_b}{P_{05}}\right) \tag{4}$$

$$\dot{m}_g = (1+f)\dot{m}_a \tag{5}$$

$$f_{th} = \eta_b f \tag{6}$$

$$f_{th} = \frac{c_{pg}T_{03} - c_{pa}T_{05}}{LHV - c_{pg}T_{03}} \tag{7}$$

2.1.3. Radial Inflow Turbine

The work produced by the pressure expansion ratio [18] across the turbine is shown in Eqs. (8) and (9). The turbine-generated power is calculated in Eq. (10).

$$\pi_t = \frac{P_{03}}{P_{04}} \tag{8}$$

$$\frac{T_{04}}{T_{03}}=1-\eta_t\left(1-\pi_t^{\frac{\gamma_g-1}{\gamma_g}}\right) \tag{9}$$

$$\dot{w}_t=\dot{m}_g(1+f)c_{pg}T_{03}\left(1-\frac{P_{04}}{P_{03}}\right). \tag{10}$$

2.1.4. Recuperator

The correlations between the pressure, pressure drop across the recuperator, temperature, and recuperator effectiveness are expressed in Eqs. (11)–(13). The heat recovery at the recuperator can be calculated as Eq. (14).

$$\frac{P_{04}}{P_{06}}=\frac{1}{1-\frac{\Delta P_{rg}}{P_{04}}} \tag{11}$$

$$\frac{P_{02}}{P_{05}}=\frac{1}{1-\frac{\Delta P_{ra}}{P_{04}}} \tag{12}$$

$$\epsilon_r=\frac{T_{05}-T_{02}}{T_{04}-T_{02}} \tag{13}$$

$$\dot{q}_r=\dot{m}_a c_{pa}(T_{05}-T_{02}) \tag{14}$$

The work output of the micro gas turbine and its electrical efficiency can be expressed as follows:

$$\dot{w}_{gen}=\eta_{gen}\left(\dot{w}_t-\frac{1}{\eta_m}\dot{w}_c\right) \tag{15}$$

$$\eta_e=\frac{\dot{w}_{gen}}{q_f} \tag{16}$$

2.2. Part-Load Performance and Model Validation

The compressor and turbine characteristic maps are needed to analyze the performance of the micro gas turbine. For practical purposes, characteristic maps from the manufacturer provide us with the micro-gas-turbine performance. However, for design purposes or if maps are not accessible, characteristic maps can be developed using a scaling method.

2.2.1. Design-Point Performance of the Micro Gasturbine

With the help of the thermodynamics in the previous section, the design-point parameters that contain all the information about the system behavior can be calculated; the micro gas turbine is 30-kW rated. The design-point parameters meeting the ISO design conditions (288.15 K, 101.325 kPa) and the design specification data of the Capstone C30 from the manufacturing datasheet [9, 15] are shown for comparison in Table 1.

2.2.2. Off-Design-Point Performance of the Micro Gas Turbine

A micro gas turbine normally faces sudden changes of load or ambient conditions, or even inconsistent calorific values. This results in off-design point operation, and the micro gas turbine yields a part-load performance. To analyze the

part-load performance of a micro gas turbine, the compressor and turbine performance formulas were applied, together with the reduced-parameter concept and a modified Flügel formula. The corrected rotational speed of the compressor is defined as:

Table 1: 30-kW Micro-Gas-Turbine Design Specifications

Description	Capstone C30	Design-Point Parameter	Unit
Micro-gas-turbine electric power	30	30	kW
Micro-gas-turbine exit temperature	872	868	K
Ambient temperature	288	288	K
Turbine inlet temperature	1114	1114	K
Ambient pressure	0.11013	0.101325	MPa
Compressor pressure ratio	3.6	3.6	-
Recuperator effectiveness	0.79	0.80	-
Combustion chamber efficiency	0.98	0.98	-
Compressor efficiency	0.78	0.75	-
Turbine efficiency	0.83	0.80	-
Generator efficiency	0.98	0.98	-

$$n_c = \frac{N}{\sqrt{T_{01}}},. \tag{17}$$

given that

$$\Gamma_c = \frac{\dot{m}_a\sqrt{T_{01}}}{P_{01}} \tag{18}$$

The compressor pressure ratio and its efficiency are shown as a function of the corrected mass flow-rate and specific speed. The compressor performance characteristic formulas are as follows.

$$\bar{\pi}_c = c_1\bar{\Gamma}_c^{\,2} + c_2\bar{\Gamma}_c + c_3 \tag{19}$$

$$\bar{\eta}_c = \frac{\bar{n}_c\left[1-c_4\left(1-\bar{n}_c\right)^2\right]}{\bar{\Gamma}_c}\left(2-\frac{\bar{n}_c}{\bar{\Gamma}_c}\right), \tag{20}$$

where

$$c_1 = \frac{\bar{n}_c}{\left[\left(p\left(1-\frac{m}{\bar{n}_c}\right)\right)+\bar{n}_c\left(\bar{n}_c-m\right)^2\right]},\ c_2 = \frac{\left(p-2m\bar{n}_c^{\,2}\right)}{\left[\left(p\left(1-\frac{m}{\bar{n}_c}\right)\right)+\bar{n}_c\left(\bar{n}_c-m\right)^2\right]},\ c_3 = \frac{-\left(pm\bar{n}_c-m^2\bar{n}_c^{\,3}\right)}{\left[\left(p\left(1-\frac{m}{\bar{n}_c}\right)\right)+\bar{n}_c\left(\bar{n}_c-m\right)^2\right]}$$

Variables m and p are the shape factor and position factor, respectively, and are defined as below. Values of m = 1.8 and p = 1.8 were selected.

$$\sqrt[3]{p} \geq \frac{2m}{3}$$

The corrected rotational speed of the turbine is defined as

$$n_t=\frac{N}{\sqrt{T_{03}}}\ ,\ \Gamma_t=\frac{\dot{m}_g\sqrt{T_{03}}}{P_{03}} \tag{21}$$

Similar to the compressor, the turbine expansion ratio and its efficiency are shown as a function of the corrected mass flow-rate and specific speed. The turbine performance-characteristic formulas are as follows:

$$\bar{\Gamma}_t=\alpha\sqrt{\frac{T_{03}}{T_3}}\sqrt{\frac{(\pi_t^2-1)}{(\pi_{t0}^2-1)}}\ ,\qquad \alpha=\sqrt{1.4-\frac{0.4n_t}{n_{0t}}} \tag{22}$$

$$\bar{\eta}_t=\left[1-t\left(1-\bar{n}_c\right)^2\right]\left(\frac{\bar{n}_t}{\bar{\Gamma}_t}\right)\left(2-\frac{\bar{n}_t}{\bar{\Gamma}_t}\right) \tag{23}$$

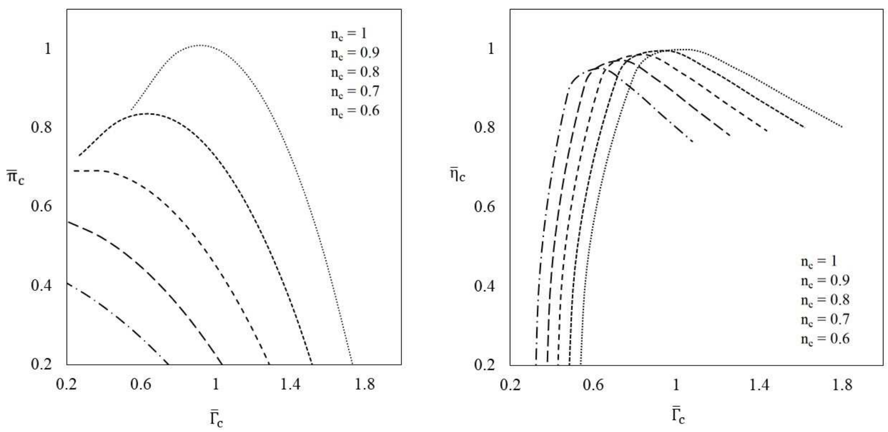

Figure 2: Compressor Performance Characteristic Map

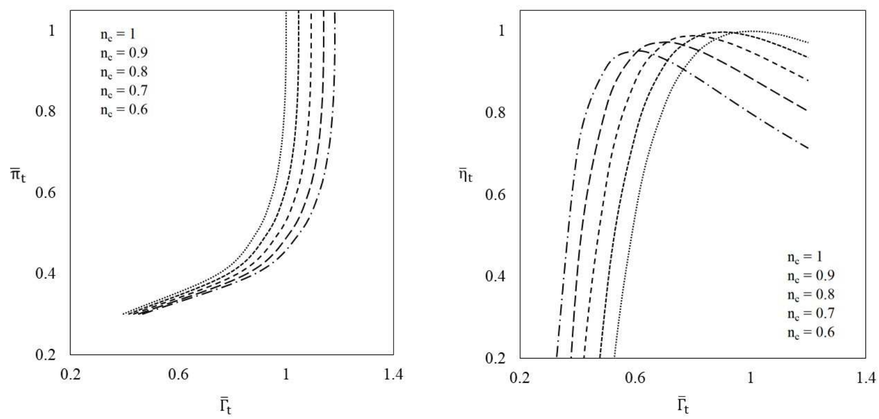

Figure 3: Turbine Performance Characteristic Map

The compressor and turbine performance formulas, Eqs. (17)–(23), were manipulated to analyze the part-load performance of a micro gas turbine. The compressor characteristic maps and turbine characteristic maps were completed and are shown in Figure 2 and Figure 3, respectively. The design-point parameter can be found at the intersecting point where the pressure ratio, efficiency, and correcting speed are equal to 1. The part-load performance of a micro gas turbine is clarified when the correcting speeds are varied from 1, 0.9, 0.8, 0.7, and 0.6. The pressure ratio and mass flow rate also decrease with the reduced correcting speed.

3. MODEL VALIDATION

The model from the previous section was validated against the Capstone C30 manufacturer data, available in the open literature, the results from testing a Capstone 30-kW, summarized, and the analytical results shown in [15]. The developed-model calculations were carried out for ISO design conditions. Natural gas properties were applied for model-validation purposes. The electrical efficiency of a 30-kW recuperated micro gas turbine, operating at part-load conditions, is shown in comparison with three different approaches; see Figure 4. It can be seen from the figure that the electrical efficiencies of the developed model are similar to the testing results, though they are slightly lower than those obtained from the manufacturing datasheet and analytical method. However, the trend of the electrical efficiencies was the same.

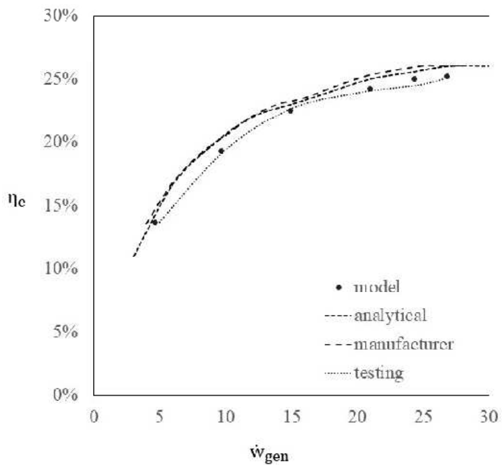

Figure 4: Electrical Efficiency of 30-kW-Micro Gas Turbine

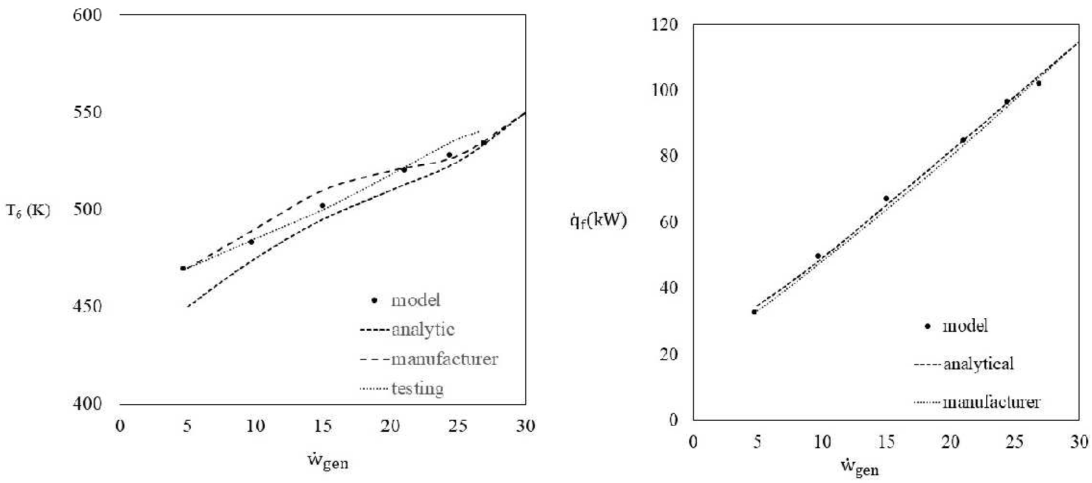

Figure 5: Exhaust Gas Temperature and Fuel Energy Rate of 30-kW-Micro Gas Turbine

The exhaust gas temperature and the fuel energy rate of a micro gas turbine are presented in Figure 5. According to the figure, the exhaust gas temperature of the developed model was close to the testing results and lay between the manufacturing data and the analytical results. The fuel energy rate from the developed model was also in good agreement and marginally higher than the manufacturing data and analytical results. The results from the developed model followed the tendencies of the manufacturing data, testing results, and analytical results. Therefore, the developed model was used to analyze the performance throughout the research.

4. RESULTS AND DISCUSSIONS

To investigate the influence of the fuel on the micro-gas-turbine performance with various electrical-load demands, three types of fuel—natural gas, ethanol from sugarcane, and biogas from palm-oil residues were selected. The lower heating values (LHV) for each type of fuel were applied to the developed model. The LHVs of ethanol from sugarcane and biogas from palm-oil residues are approximately 29,950 kJ/kg and 18,260 kJ/kg, respectively. Because the LHVs of ethanol and biogas are rather small, compared to natural gas, a larger quantity of ethanol and biogas must be fed into the system to maintain a constant power output, as shown in Figure 6. In other words, to maintain the power output of a micro gas turbine, it is necessary to maintain the heat input at the combustion chamber, which means that larger quantities of ethanol and biogas are needed. In the power-load demand range of 5 kW to 27 kW, the ethanol consumption rate changes from 47.6% to 50.9%, while the biogas consumption rate changes from 143.8% to 149.9%, compared to natural gas.

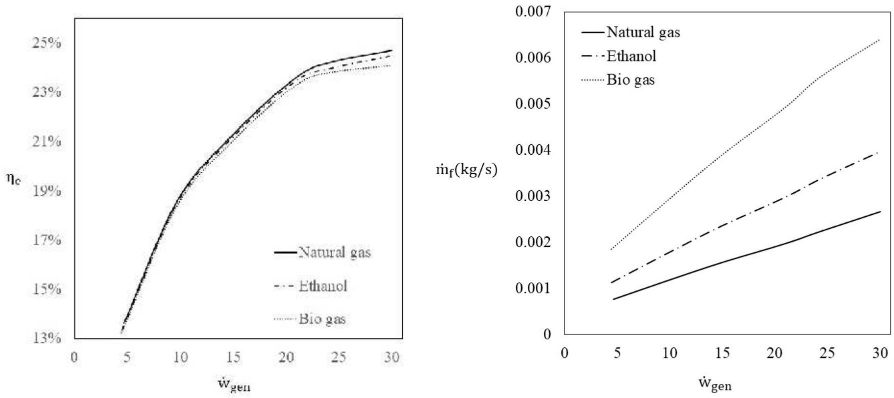

Figure 6: Effects of Fuel Variation on the Electrical Efficiency, Fuel Consumption, and Power Output of Micro Gas Turbine

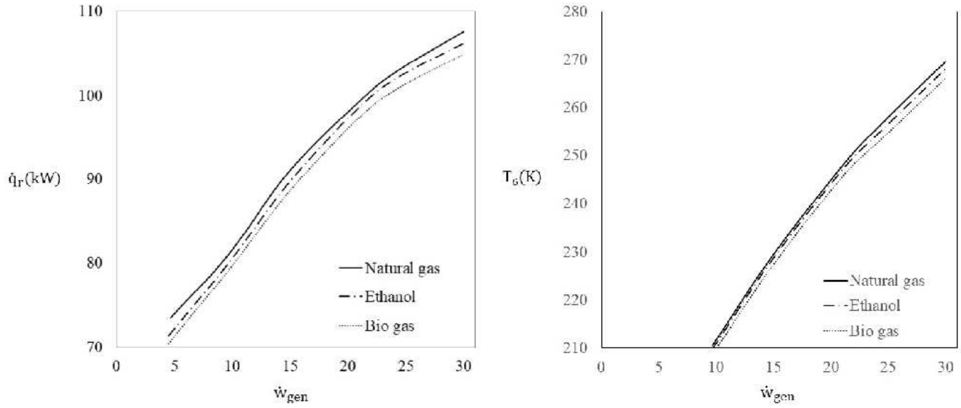

Figure 7: Effects of Fuel Variation on Heat Recovery at Recuperator, Exhaustt Emperature, and Power Output of Micro Gas Turbine

The quantity of fuel that is fed into the micro gas turbine also affects the heat recovery at the recuperator. The heat recovery at the recuperator decreases when the micro gas turbine operates with ethanol or biogas, instead of natural gas. As shown in Figure 7, the heat-recovery percentages for each fuel type are constant throughout the range of the power-load demand. The effects of the fuel variationon the exhaust temperature and power output of the micro gas turbine are shown. The results show that the exhaust temperatures for all fuel types linearly increase as the power output increases.

Similar to the heat recovery, the change percentages in the exhaust temperature for each fuel type are constant throughout the range of the power-load demand.

5. CONCLUSIONS

This study was undertaken to evaluate the performance of a 30-kW micro gas turbine. A centrifugal compressor and radial inflow turbine were designed for a 30-kW micro gas turbine. The compressor performance characteristic maps and turbine performance characteristic maps were created to clarify the part-load performance of the micro gas turbine. A steady-state thermodynamic model of the micro gas turbine was developed and validated against the Capstone C30 manufacturing data, results from testing a Capstone C30, and analytical results. The propertiesof natural gas were applied for model-validation purposes, and the results were found to be in good agreement. Natural gas, ethanol from sugarcane, and biogas from palm-oil residues were selected to investigate their effect on micro-gas-turbine performance. The most obvious finding was no significant difference in the micro-gas-turbine performance when the fuel changed from natural gas to renewable energy; i.e., ethanol and biogas. One implication of this is the possibility of utilizing renewable energy instead of natural gas, which also benefits the environment. However, from an economic point of view, using renewable energy instead of natural gas requires a larger amount of fuel to be fed into the system, which increases the fuel expenses and operational costs. The present study lays essential groundwork for the design and development of a micro gas turbine that uses renewable energy sources. The findings of this study have a number of important implications for future practice. For example, a further study could assess the long-term positive and negative effects of switching to renewable energy sources. Investigation and experimentation on a large-scale biofuel micro gas turbine are also strongly recommended.

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APPENDIX-1

Nomenclature		Subscript and Superscript	
f	fuel-air ratio	b	combustion chamber
LHV	lower heating value	c	compressor
m,p,t	shape factor	cc	cold-side recuperator
N	rotational speed	e	electrical
n	corrected rotational speed	f	fuel
V	volume flow rate	m	mechanical
Greek letter		f	r
π	pressure ratio	s	specific
η	efficiency	th	theoretical parameter
γ	specific heat ratio	0	design value
ϵ	effectiveness	1,2,3,4,5,6	state
Γ	correcting mass flow rate	·	rate
		-	divided by design value